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A Capillary Tube-Refrigerant Charge Design Methodology for Household Refrigerators-Part II: Equivalent Diameter and Test Procedure

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A Capillary Tube - Refrigerant Charge Design Methodology for Household Refrigerators – Part II: Equivalent Diameter and Test Procedure

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ABSTRACT

In the first part of this work an experimental apparatus was designed and constructed to map the energy consumption of a household refrigerator subjected to several combinations of refrigerant charge and expansion restriction. In the second part, the expansion restriction imposed by the pair metering valve-capillary tube was converted into an equivalent tube diameter applying two different procedures: dry nitrogen flow and mathematical modeling. An empirical correlation to estimate the energy consumption based on the capillary tube inner diameter and refrigerant charge was also developed and used during the minimization process. Different strategies were also explored in order to reduce the amount of experiments to a minimum. It was found that at least 14 data points, collected with three different refrigerant charges, are required to ensure the convergence of the energy consumption minimization process.

1. INTRODUCTION

Refrigeration and air conditioning appliances account for around 50% of the energy consumed by the residential sector in Brazil and this represents approximately 10% of the national electricity production. This high fraction is due to the large amounts of units being used and also to their low thermodynamic efficiencies.

It is well known that the energy consumed by household refrigerators is dependent on each of the components and also on the refrigerant charge and ambient conditions. The lowest-cost component of a household refrigerator is the capillary tube, which implies that attempts to improve performance should focus primarily on this component. However, each capillary requires a distinct refrigerant charge and vice-versa (Vjacheslav et al., 2001).

Björk and Palm (2006) conducted a parametric analysis with a household refrigerator working under cyclic conditions in an attempt to identify the ideal combination of expansion restriction and refrigerant charge, without employing the commonly used trial-and-error procedure. Based on a database of 600 data points, comprised of several combinations of expansion restriction, refrigerant charge and ambient temperature, the authors were able to identify a wide region of minimum energy consumption. More recently, Boeng and Melo (2012) carried out a parametric analysis similar to that of Björk and Palm (2006) but using a steady-state energy consumption measurement methodology (Hermes et al., 2012) and distinct automated procedures for varying the refrigerant charge and expansion restriction.

In this work the restriction provided by the pair metering valve-capillary tube was firstly converted into an equivalent capillary tube and an energy consumption minimization procedure was then developed. Finally, an experimental strategy was identified to ensure the convergence of the energy consumption minimization process based on a reduced amount of experimental data.

2. EQUIVALENT CAPILLARY TUBE

The expansion restriction was varied through the opening of the metering valve. The valve opening was associated with the remaining turns for the total closure. The fully-open position corresponds to the 9.75 opening, while the fully-closed position corresponds to the zero opening. As the refrigerant mass flow rate does not vary linearly with the valve opening, the number of turns was correlated with a position scale, varying between 0 and 11. The opening-position relationship, obtained applying variations in the mass flow rate of around 0.04 kg/h, is shown in Table 1.

Table 1: Metering valve opening vs. position

Opening [turns]	9.75	1.750	1.000	0.750	0.650	0.600	0.550	0.525	0.500	0.475	0.450	0.425
Position [-]	11	10	9	8	7	6	5	4	3	2	1	0

The combination of expansion restriction with refrigerant charge resulted in 95 experimental data points, based on which the energy consumption contour map, shown in Figure 1, was plotted. This figure is similar to that obtained by Björk and Palm (2006), suggesting a common pattern for different household refrigerators. It was observed that the refrigerator under study runs with a minimum energy consumption of the order of 50kWh/month when subjected to various combinations of restriction and charge. It is worth noting that an increased restriction requires an increased charge and vice-versa.

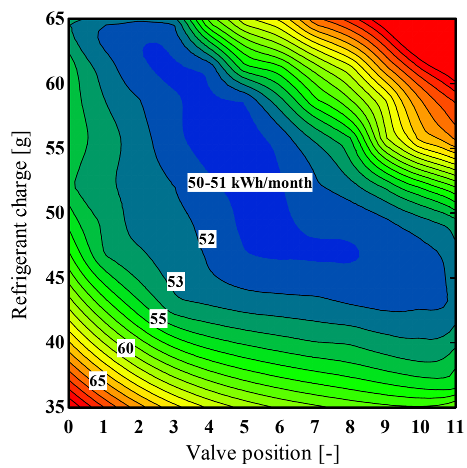

Figure 1: Energy consumption vs. refrigerant charge and expansion restriction

The restriction data used in the parametric analysis, expressed in number of turns or position, were converted into an equivalent capillary tube, *i.e.*, a capillary tube which supplies the same mass flow rate as the pair metering valve-capillary tube. To this end a dry nitrogen flow testing apparatus was firstly used and later the results thus obtained were compared to the predictions of a capillary tube model.

2.1. Dry nitrogen flow

The restriction imposed by a capillary tube is essentially dependent on its geometry, especially the internal diameter of the tube. Kipp and Schmidt (1961) carried out experiments with capillary tubes of different geometries, *i.e.*, diameter and length, using dry nitrogen as the working fluid. They correlated the dry nitrogen volumetric flow, $\dot{V}_{N_2}$, with the tube diameter, D, and length, L, and with the inlet pressure, p_{in}. The correlation has the following form:

$$\dot{V}_{N_2} = c_1 L^{-c_2} D^{c_3} \sqrt{{p_{in}}^2 - 1} \tag{1}$$

where, $c_1 = 2.5$, $c_2 = 0.5$ and $c_3 = 2.5$

The Kipps and Schmidt (1961) correlation is widely used since it allows the determination of the size of the capillary tube from a known nitrogen flow. In this study, this expression was used to calculate the inner diameter of the equivalent capillary tube, as indicated below,

$$D = \left(\frac{\dot{V}_{N_2} L^{c_2}}{c_1\sqrt{{p_{in}}^2 - 1}}\right)^{\frac{1}{c_3}} \tag{2}$$

To this end, the nitrogen flow though the pair metering valve-capillary tube was measured using a test rig constructed according to the specifications of the ASHRAE 28 (1996) standard. Firstly, 28 tests were carried out with capillary tubes of different geometries (diameter of 0.64 mm to 1.07 mm and length of 1 m to 3 m), and it was found that the Kipps and Schmidt (1961) correlation predicted 65% of the experimental data within an error band of ± 10.0%. In order to further improve the calculation process the empirical coefficients of Equation (2) were recalculated and the following values were obtained: $c_1 = 2.362$, $c_2 = 0.496$ e $c_3 = 2.657$. Secondly, the mass flow rate through the pair metering valve-capillary tube was measured for all predetermined valve positions, using an inlet pressure of 7.89 bar (ASHRAE 28, 1996). The data thus obtained were subsequently converted to the equivalent diameter applying Equation (2), as shown in Table 2.

Table 2: Equivalent internal diameter vs. valve position (*VP*)

experimental	**VP**	**[-]**	11	10	9	8	7	6	5	4	3	2	1	0
	D	**[mm]**	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83
	L	**[m]**	3.32	3.32	3.32	3.32	3.32	3.32	3.32	3.32	3.32	3.32	3.32	3.32
	$V_{N_2,exp}$	**[l/min]**	6.23	5.71	4.87	4.30	3.91	3.79	3.64	3.53	3.33	3.17	2.99	2.85
correlation	**D_{N_2}**	**[mm]**	0.83	0.81	0.76	0.73	0.71	0.70	0.68	0.67	0.66	0.65	0.63	0.62

2.2. Mathematical modeling

The equivalent tube diameter was also calculated through a mathematic model for non-adiabatic capillary tubes, in order to corroborate the results provided by the nitrogen flow procedure. This is a distributed model, which means that the conservation equations are applied to a series of infinitesimal control volumes distributed along the tube length (Hermes et al., 2008). Based on the simplifications described by Hermes et al. (2008), the conservative equations applied to a fluid element of length *dz* can then be written as follows:

$$\frac{dG}{dz} = 0 \tag{3}$$

$$G\frac{dV}{dz} + \frac{dp}{dz} + \sigma\frac{4}{D} = 0 \tag{4}$$

where G is the refrigerant mass flux $[kg.s^{-1}.m^{-2}]$, V is the average flow axial velocity $[m.s^{-1}]$, p is the pressure $[Pa]$, $\tau = fGV/8$ is the shear stress at the tube wall $[Pa]$, f represents the Darcy friction factor, D is the tube inner diameter and h represents the specific enthalpy $[J.kg^{-1}]$.

Considering that the mass flux is defined as $G = V/v$, where v is the specific volume $[m^3.kg^{-1}]$, it can be easily shown that $dV = Gdv$. Replacing dV and V in Equation (4) and defining dv from the Maxwell relations (Hermes *et al.* (2008), yields:

$$\frac{dz}{dp} = -\frac{D}{4}\frac{1 + G^2\left[v\left(\frac{\partial v}{\partial h}\right)_p + \left(\frac{\partial v}{\partial p}\right)_h\right]}{\tau\left[1 + G^2 v\left(\frac{\partial v}{\partial h}\right)_p\right] + qG\left(\frac{\partial v}{\partial h}\right)_p} \tag{5}$$

Applying Equation (5) to the adiabatic region ($q = 0$), where the flow is considered isenthalpic ($\partial v/\partial h = 0$) and introducing the definition of τ, yields:

$$\frac{dz}{dp} = -\frac{2D}{fG^2 v}\left[1 + G^2\left(\frac{\partial v}{\partial p}\right)_h\right] \tag{6}$$

Yilmaz and Unal (1996) proposed a mathematical model for calculating the refrigerant mass flow rate through adiabatic capillary tubes assuming an isenthalpic flow. The same approach was adopted in this study since the effect of the kinetic energy variation on the specific enthalpy can be neglected in the non-adiabatic region (Hermes et al., 2008). In this region the enthalpy variation can thus be expressed only through the heat exchanges with the suction line, as shown in Equation (7).

$$\dot{m}dh = -qP_{er}dz \tag{7}$$

where $P_{er} = \pi D$ is the capillary tube perimeter $[m]$ and $\dot{m} = G\pi D^2/4$ is the mass flow rate $[kg/s]$. The energy conservation equation can then be written simply as:

$$Gdh = -\frac{4q}{D}dz \tag{8}$$

Considering that only superheated vapor flows in the suction line ($c_p = \frac{dh}{dT}$):

$$Gc_{ps}dT_s = \frac{4q}{D}dz \tag{9}$$

Equations (5), (8) and (9) are all differential first-order equations and therefore only one boundary condition is required for each of them. The boundary conditions are defined by the refrigerant thermodynamic state at the inlet of the capillary tube (condensation temperature and pressure) and at the inlet of the suction line (evaporation temperature and pressure). It should be noted that there are four boundary conditions for only three equations. The extra boundary condition – the evaporation or the choked pressure – is used to calculate the mass flow rate or the internal diameter (Melo et al., 1992). Choked flow occurs when the Fanno, *ds/dh* → 0 (Stoecker and Jones, 1983), or the Fauske (1962), *dP/dz* → - ∞, criterion, which are mathematically equivalent (Silva, 2008), applies.

To simultaneously solve the set of equations, the temperature at the outlet of the suction line must firstly be arbitrated, and successively corrected according to the difference between the actual and the calculated values for the temperature at the inlet of the suction line. This iterative procedure undermines the convergence process, creating instabilities in the model and increasing the computational time. In order to avoid this iterative procedure Hermes et al. (2008) introduced a new approach to modeling the heat exchange in the non-adiabatic region of the capillary tube, considering this region as a counter-flow heat exchanger. To this end, the ε-NUT method for fluids of equivalent thermal capacities was used (Incropera and DeWitt, 2011). In fact, this was an approximation since the ε-NUT method only applies to heat exchangers without phase change. However, the inherent error caused by this approximation did not significantly affect the results.

The numerical model was validated against the experimental data for HC-600a and HFC-134a collected by Melo et al., (2002) and Zangari, (1998). This experimental database is comprised of different geometries (internal diameter of 0.553 mm to 0.830 mm, length of 3m to 4m) and operating conditions (condensation and evaporation pressures, superheating and subcooling degrees). Figure 2 compares the refrigerant mass flow rates predicted by the model with the experimental data. As can be seen the numerical model predicts 86% of the experimental data within an error band of ± 10%.

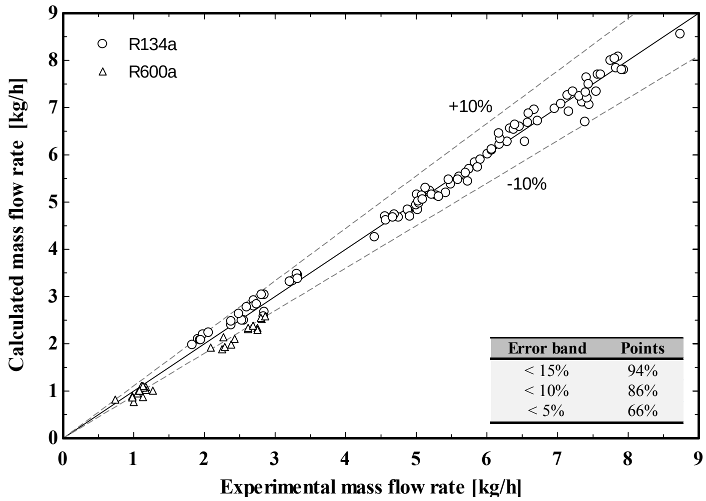

Figure 2: Model predictions vs. experimental data of Melo *et al.* (2002) and Zangari (1998)

Some of the experiments were carried out with the metering valve completely open, meaning that the flow restriction was governed only by a 0.83 mm diameter and 3.32 m length capillary tube. This allowed a further comparison between the model predictions ($\dot{m}_{calc}$) and the experimental data ($\dot{m}_{exp}$) collected for the actual refrigerator. This comparison is shown in Table 3.

Table 3: Experimental vs. calculated mass flow rate

Charge	**[g]**	36.7	37.7	40.5	42.9	46.9	49.3	52.1	55.7	58.7	61.3	64.7
ΔT_{sub}	**[°C]**	2.7	2.8	3.1	3.2	3.3	3.3	3.4	3.9	4.4	4.9	5.3
m_{exp}	**[kg/h]**	1.21	1.24	1.30	1.34	1.38	1.42	1.51	1.65	1.85	1.98	2.15
m_{calc}	**[kg/h]**	2.69	2.72	2.76	2.78	2.81	2.80	2.77	2.78	2.85	2.87	2.94

It can be noted that the deviations were between 36% and 122%, depending on the subcooling (ΔT_{sub}). This discrepancy is much higher than expected since the numerical model was previously validated (see Figure 2), all measuring devices were calibrated and the tests were carried out with extreme care. Since the discrepancy increases with decreasing subcooling, additional experiments were carried out in order to check for the flow pattern at the entrance of the capillary tube. To this end a transparent filter dryer was installed at the inlet of the capillary tube.

Figure 3 shows three pictures of the flow pattern at the inlet of the capillary tube during a test carried out with a refrigerant charge of 64.7 g, with the metering valve fully open and with a subcooling of 5.3 ^{o}C. These pictures were taken with a high-speed camera at different times. It can be clearly seen that the flow is intermittent, alternating between slug and bubbly flows, even with the system running under steady-state conditions. Therefore, even with 5^{o}C of subcooling, some vapor bubbles are present at the entrance of the capillary, the fluid being a non-equilibrium mixture of subcooled liquid and saturated vapor, instead of being purely liquid.

A simulation exercise was then carried out in order to find the refrigerant quality at the inlet of the capillary which matches the predicted and measured mass flow rates. Only the experimental data with the valve fully open were used in this exercise since the tube diameter was used as an input parameter. The results are shown in Table 4.

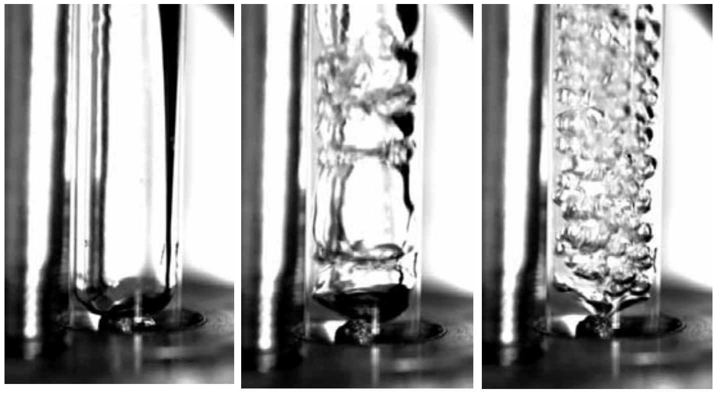

Figure 3: Visualization of capillary tube inlet flow for 64.7g of charge

Table 4: Capillary tube inlet quality prediction

Charge	[g]	36.7	37.7	40.5	42.9	46.9	49.3	52.1	55.7	58.7	61.3	64.7
ΔT_{sub}	[°C]	2.7	2.8	3.1	3.2	3.3	3.3	3.4	3.9	4.4	4.9	5.3
m	[kg/h]	1.21	1.24	1.30	1.34	1.38	1.42	1.51	1.65	1.85	1.98	2.15
x	[%]	11.8	11.5	10.6	10.1	9.7	9.3	7.9	6.4	5.0	4.0	3.2

An empirical correlation to predict the refrigerant quality at the inlet of the capillary tube (χ), based on the data shown in Table (4), was then developed and is presented below:

$$\chi = \frac{\dot{m} \cdot 98.884 \cdot \Delta T_{sub}{}^{-2.015}}{0.368 \cdot \Delta T_{sub} + 0.195} \tag{10}$$

Equation (10) was then applied to the entire database gathered by Boeng and Melo (2012), with encompasses refrigerant charges from 36.7g to 64.7g and 12 valve openings. In this way, the equivalent tube diameter was calculated for all conditions, as illustrated in Figure 4. The mean values for each opening are shown on the right-hand side of Figure 4. It can be noted that the method adopted is quite satisfactory, especially for large openings. A significant scattering was found in the case of smaller openings, most likely due to the growth of the vapor mass fraction at the inlet of the capillary.

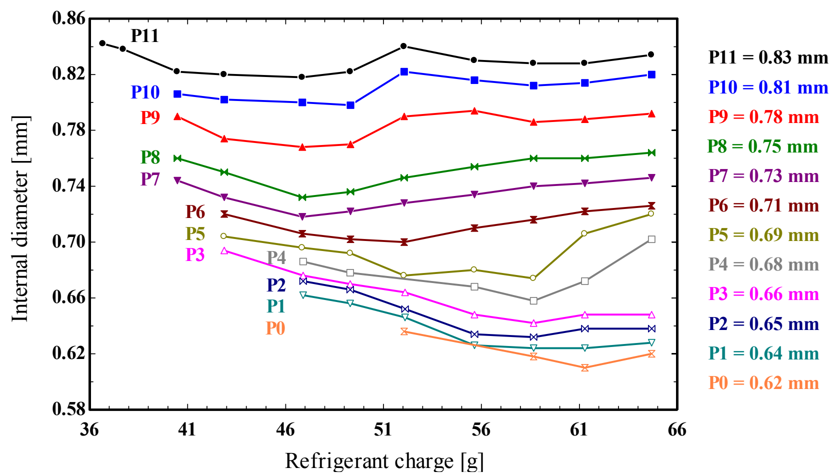

Figure 4: Equivalent internal diameter for all experimental tests

2.3. Comparative analysis

Table 5 shows a comparison between the equivalent capillary tube inner diameters calculated from the nitrogen flow measurements and those of the mathematical model. It is worth noting that the results are almost coincident, with only small deviations of the order of ±3%.

Table 5: Comparison: nitrogen flow vs. mathematical model

Position	[-]	11	10	9	8	7	6	5	4	3	2	1	0
N2 flow	[mm]	0.83	0.81	0.76	0.73	0.71	0.70	0.68	0.67	0.66	0.65	0.63	0.62
Model	[mm]	0.83	0.81	0.78	0.75	0.73	0.71	0.69	0.68	0.66	0.65	0.64	0.62

3. MINIMIZATION METHOD

The experimental tests resulted in a contour map of the energy consumption as a function of the refrigerant charge and capillary tube internal diameter, which was quite similar to that obtained by Björk and Palm (2006) who worked with a different household refrigerator.

3.1. Empirical correlation

An empirical correlation to estimate the energy consumption of a household refrigerator based on the inner diameter of the capillary tube (D) and refrigerant charge (C_{fr}) was developed using the TableCurve 3D software (Khader, 2002). The option chosen for application was a relatively simple correlation with few parameters and a high coefficient of determination (R^2). Equation (11) was thus selected, with the empirical parameters given in Table 6. Figures 5 shows this equation fitted to all experimental data.

$$EC = c_1 + c_2 D + c_3 C_{fr} + c_4 D^2 + c_5 {C_{fr}}^2 + c_6 D C_{fr} \tag{11}$$

Table 6: Equation (6) empirical parameters

c_1	c_2	c_3	c_4	c_5	c_6
641.62	-9.8728	-972.11	0.045827	424.68	7.4451

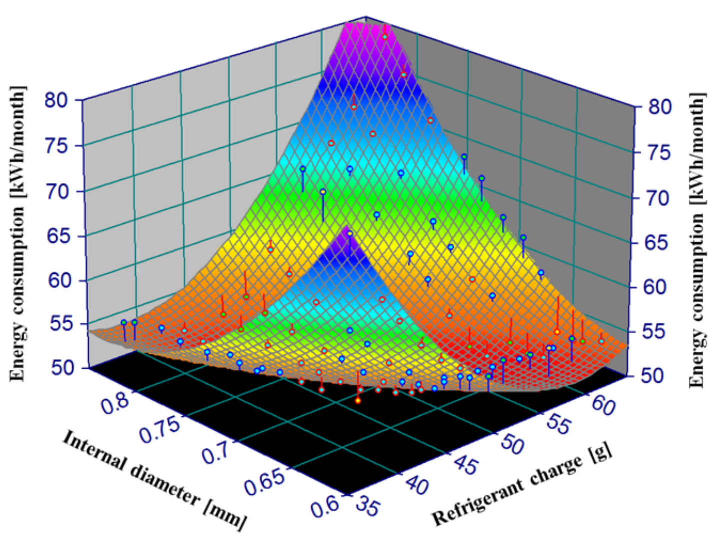

Figure 5: Empirical correlation for energy consumption prediction

3.2. Minimum energy consumption

The pair internal diameter-refrigerant charge, which minimizes the energy consumption, can be determined through the EES software (Klein and Alvarado, 2004) applied to Equation (11). To this end, the variable metric method for two degrees of freedom was used. The minimization process resulted in an energy consumption of 50.7 kWh/month, a refrigerant charge of 51.2 g and an internal diameter of 0.70 mm, as illustrated in Figure 6. Coincidentally, the refrigerator used in this study was originally assembled with a 0.70 mm (I. D.) capillary tube and charged with 47 g of refrigerant. Although this is not exactly the ideal match, it may be concluded that the refrigerator investigated in the current study is well balanced in terms of capillary and charge, a situation distinct from that prevailing in the market.

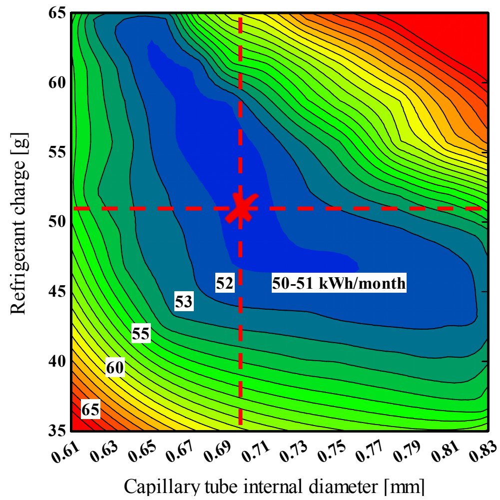

Figure 6: Ideal point of operation

3.3. Test procedure

A test procedure was also developed in order to avoid unnecessary experimental effort. Within this context, different testing strategies were applied to the 95 data points collected in this study. To this end, the scanning process and the number of data points within each scan were both varied. For each case the empirical coefficients of Equation (11) were re-fitted and a new optimization process carried out. The process was considered convergent when the resulting energy consumption fell within the optimum region. The following strategies were investigated (see Figure 7): (a) random tests, (b) fixed charge and restriction, (c) variable charge with 2 or 3 series of fixed restriction, and (d) variable restriction with 2 or 3 series of fixed charges.

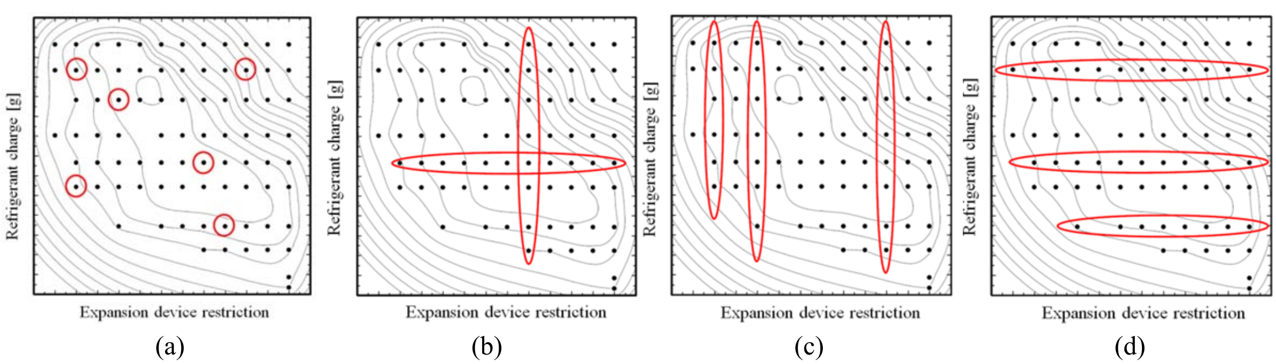

Figure 7: Scheme of the test strategies

Firstly, the random test strategy was considered. Different sets of experimental data were thus chosen from the array of data available. It was found that the minimization process converged when ten or more data points were used. However, in practical terms, this strategy is limited since it requires a considerable length of time to vary both the charge and the restriction. Secondly, the restriction was varied while the charge was maintained fixed and *vice versa*. It was found that with this strategy the minimization process diverged, regardless of the number of data points used. Thirdly, the charge was varied while the restriction was kept fixed at three different values. In this particular case the convergence was only partial and this testing strategy was subsequently discarded. Finally, the charge was kept fixed at different values while the expansion was varied. It was observed that with this strategy the minimization process was divergent with two different charges but unconditionally convergent with three different charges. On the basis of these findings it can be concluded that only the strategies involving random tests and the restriction variation with three charge values are suitable for the energy consumption minimization process.

3.4. Minimum amount of data

The testing strategy option based on the restriction variation with three charge values was selected since it is much faster and easier to apply than that based on random tests. The next step was to determine the minimum amount of data points to assure the convergence of the minimization process. For this purpose four sets of data were randomly chosen each with 10, 14, 18, 22, 26 and 30 data points distributed among three distinct charge values. It was found that the minimization process was unconditionally convergent if 14 or more data points were used, and this number of data points is thus recommended herein. Considering the experimental procedure adopted in this study it may be estimated that at least 10 days are required to collect the minimum amount of data for the energy minimization process. The test procedure described in this section was also applied to the database of Björk and Palm (2006) and similar results were obtained.

4. CONCLUSIONS

This aim of this study was to develop a methodology to minimize the energy consumption of household refrigerators focused on the proper choice of the pair capillary tube-refrigerant charge. The energy consumption was firstly mapped, by varying the expansion restriction and the refrigerant charge. The restriction imposed by the pair metering valve-capillary tube was then converted into an equivalent tube diameter through two different procedures that provided almost the same results. The energy consumption was correlated with the tube diameter and refrigerant charge and minimized through the variable metric method. Finally, a test procedure to reduce the amount of experimental data required for the minimization process was also developed. The main conclusions of this work are summarized below.

- There is a strong similarity between the experimental data of this study and those of Bjork and Palm (2006), despite the dissimilarity between the refrigerators. This suggests that a common behavior for most refrigerators may exist;
- The refrigerator under study is charged with 47g of HC-600a, assembled with a 0.70 mm (I. D.) capillary tube and consumes 50.9 kWh/month. The methodology proposed herein suggests a 0.70 mm (I. D.) capillary tube, a refrigerant charge of 51.2 g and an energy consumption of 50.7 kWh/month. It may thus be concluded that this particular product is well balanced in terms of capillary tube and refrigerant charge;
- The pictures captured by a high-speed camera at the entrance of the capillary tube revealed the presence of vapor bubbles even with some degree of subcooling, indicating a thermodynamic non-equilibrium flow comprised of subcooled liquid and saturated vapor;
- The variable metric method was used to minimize the energy consumption. This method is simple, robust and effective, correctly predicting the optimum operation point, both for the refrigerator under study and for that used by Björk and Palm (2006);
- Only 14 experimental data, collected with different restrictions and with a minimum of three refrigerant charges, are needed to assure the convergence of the energy minimization process.

REFERENCES

ASHRAE Standard 28, 1996, Method of Testing Flow Capacity of Refrigerant Capillary Tubes, *American Society of Heating, Refrigeration and Air Conditioning Engineers*, Atlanta, GA, USA.

Björk, E. , Palm, B., 2006, Performance of a Domestic Refrigerator under Influence of Varied Expansion Device Capacity, Refrigerant charge and Ambient Temperature, *International Journal of Refrigeration*, Vol. 29, Pages 789-798.

Boeng, J., Melo, C., 2012, A Capillary Tube - Refrigerant Charge Design Methodology for Household Refrigerators – Part I: Mapping the energy consumption, *VI Ibero-American Congress of Sciences and Cold Techniques*, Madrid, Spain.

Fauske, H.K., 1962, Contribution to the Theory of the Two Phase, One Component Critical Flow, *Internal Report, Argonne National Laboratory*, Argonne, IL, USA.

Hermes, C.J.L., Melo, C., Gonçalves, J.M., 2008, Modeling of Non-Adiabatic Capillary Tube Flows: A Simplified Approach and Comprehensive Experimental Validation, *International Journal of Refrigeration*, 31, p. 1358-1367.

Hermes, C.J.L., Melo, C., Knabben, F.T., 2012, Alternative Test Method to Assess the Energy Consumption of Household Refrigerating Appliances, *14th International Refrigeration and Air Conditioning Conference at Purdue*, West Lafayette, IN, July 16-19, 2012.

Incropera, F.P., DeWitt, D.P., 2011, *Fundamentals of Heat and Mass Transfer*, John Wiley & Sons.

Khader, M., 2002, TableCurve 3D Version 4.0, *Systat Software Inc. (SSI)*, San Jose, California, USA.

Kipp, E., Schmidt, H., 1961, Bemessung von Kapillarrohren als Drosselorgan für Kaltesatze, *Kaltetechnik*, 13 Jahrgang, Heft 2.

Klein, S.A., Alvarado, F.L., 2004, EES – Engineering Equation Solver User's Manual, *F-Chart Software*, Middleton, WI, USA.

Melo, C., Ferreira, R.T.S., Pereira, R.H., 1992, Modeling Adiabatic Capillary Tubes: A Critical Analysis, *International Refrigeration Conference: Energy Efficiency and New Refrigerants*, West Lafayette, USA, pp. 113-122.

Melo, C., Vieira, L.A.T., Pereira, R.H., 2002, Non-adiabatic Capillary Tube Flow with Isobutane, *Applied Thermal Enginnering*, Vol. 22 (No. 14), pages 1661–1672.

Silva, D.L., 2008, *Theoretical and Experimental Analysis of the Transcritical Expansion of Carbon Dioxide Through Adiabatic Capillary Tubes*, Master Thesis, Department of Mechanical Engineering, Federal University of Santa Catarina.

Stoecker, W.F., Jones, J.W., 1982, *Refrigeration and Air Conditioning*, McGraw Hill, New York.

Vjacheslav, N, Rozhentsev, A, Wang, C-C, 2001, Rationally Based Model for Evaluating the Optimal Refrigerant Mass Charge in Refrigerating Machines, *Energy Conversion and Management*, Volume 42, Issue 18, Pages 2083-2095.

Yilmaz, T., Unal, S., 1996, General Equations for the Design of Capillary Tubes, *ASME Journal of Fluids Engineering*, Vol. 118, pg. 150-154.

Zangari, J. M., 1998, *Experimental Performance Evaluation of Capillary Tube-Suction Line Heat Exchangers of the Concentric Type*, Master Thesis, Department of Mechanical Engineering, Federal University of Santa Catarina.

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